ENGINEERING LETTER | 4

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TEMPERATURE AND ALTITUDE AFFECT FAN SELECTION

INTRODUCTION

Fan performance changes with the density of the gas being handled. Therefore, all fans are cataloged at a standard condition defined as: 70°F. air, at sea level, with a gas density of .075 lb./ft.^{3} at a barometric pressure of 29.92" Hg. At any other condition, the fan's horsepower requirement and its ability to develop pressure will vary. Therefore, when the density of the gas stream is other than the standard .075 lb./ft.^{3}, correction factors must be applied to the catalog ratings in order to select the correct fan, motor, and drive.

In addition, the maximum safe speed of a wheel, shaft, or bearing can change due to an alloy becoming too brittle or too pliable at temperatures other than 70°F. Temperature derate factors must be applied to the fan's catalog maximum safe speed to ensure against overspeed situations.

HOW TO CALCULATE ACTUAL FAN PERFORMANCE AT OTHER THAN 70 DEGREES FAHRENHEIT

As illustrated in Figure 1, a fan wheel is similar to a shovel. In a given system, it will move the same volume of air regardless of the air's weight. If a fan moves 1000 CFM at 70°F., it will also move 1000 CFM at 600°F.

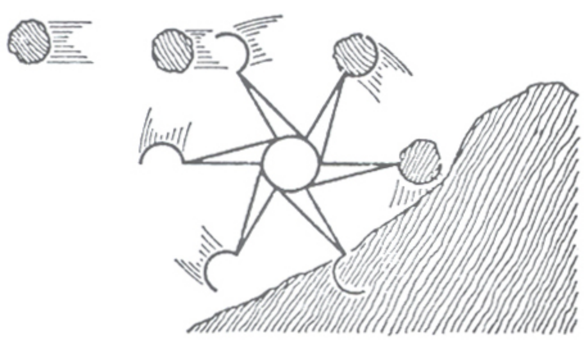

Figure 1 - A fan wheel is like a shovel.

However, air at 600°F. weighs half as much as it does at 70°F. Therefore, the fan requires just half the horsepower. (See Figure 2.) Likewise, since the air weighs half as much, it will create only half the static and velocity pressures. The reduction in static pressure is proportional to the reduction in horsepower, thus the overall fan efficiency will remain unchanged.

$$\text{Total Efficiency} = \frac{\text{CFM x Total Pressure}}{\text{6356 x Brake Horsepower}}$$

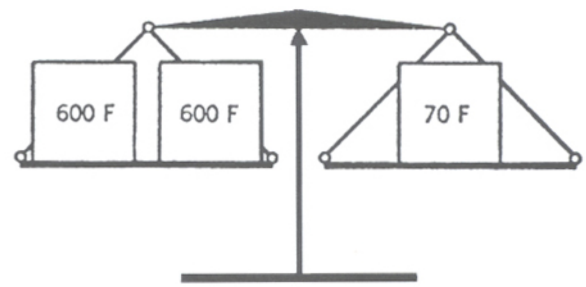

Figure 2 - With hot gas, there is less weight to shovel.

Example 1. A fan handling standard density, 70°F. air, delivers 12,400 CFM against 6" SP (static pressure) requiring 14.6 BHP (brake horsepower). If the system and fan RPM are not changed, but the inlet airstream temperature is increased to 600°F., how will the fan perform?

The fan will still deliver 12,400 CFM, but since the air at 600°F. weighs half as much as the air at 70°F., static pressure and horsepower will be cut in half. The fan will generate only 3" SP and require only 7.3 BHP.

A typical fan specification based on hot operating conditions is illustrated in Example 2.

Example 2. Required: 11,000 CFM and 6" SP at 600°F. (This means the actual, measurable static pressure of the fan at 600°F. will be 6 inches of water.)

The fan's catalog performance tables are based on 70°F. air at .075 density. The specified SP must be corrected by the ratio of the standard density to operating density. Since densities are inversely proportional to absolute temperature (degrees F. + 460):

$$6" \left(\frac{460 + 600}{460 + 70}\right) = 6" \left(\frac{1060}{530}\right) = 12"$$

The fan must be selected from the rating tables for 11,000 CFM at 12" SP. The BHP obtained from the table should be multiplied by the ratio of operating density to standard density in order to obtain the BHP at 600°F. If the rating table showed 30.0 BHP, the operating BHP would be 30.0 (530/1060) = 15.0 BHP.

In most "hot" systems, the fan is required to handle cold air until the system reaches temperature. A good example is in oven exhaust systems.

If Example 2 were such a case, the fan would require 30.0 BHP when operating at 70°F., and 15.0 BHP when the oven had warmed to 600°F. Very often a damper is furnished with the fan so that, during the warming-up period, the fan can be dampered to reduce the horsepower. Without the damper, a 30 HP motor would be needed.

Confusion can be avoided if the SP is specified at the temperature it was calculated. In Example 2, the specifications should read either:

11,000 CFM and 6" SP at 600°F., or

11,000 CFM for operation at 600°F. and 12" SP at 70°F.

Table 1 gives correction factors used to convert from a non-standard density to a standard density of 70°F. air. These factors are merely the ratios of absolute temperatures. Multiply the actual static pressure by the specific temperature/altitude factor so standard catalog rating tables can be used. Divide the brake horsepower from the catalog rating table by the temperature/altitude factor to get BHP at conditions.

Table 1 - Corrections for Temperature

Air Temperature °F.	Factor	Air Temperature °F.	Factor
-50	0.77	275	1.39
-25	0.82	300	1.43
0	0.87	325	1.48
+20	0.91	350	1.53
40	0.94	375	1.58
60	0.98	400	1.62
70	1.00	450	1.72
80	1.02	500	1.81
100	1.06	550	1.91
120	1.09	600	2.00
140	1.13	650	2.09
160	1.17	700	2.19
180	1.21	750	2.28
200	1.25	800	2.38
225	1.29	900	2.56
250	1.34	1000	2.76

Table 2 - Corrections for Altitude

Altitude Feet Above Sea Level	Factor	Altitude Feet Above Sea Level	Factor
0	1.00	5000	1.20
500	1.02	5500	1.22
1000	1.04	6000	1.25
1500	1.06	6500	1.27
2000	1.08	7000	1.30
2500	1.10	7500	1.32
3000	1.12	8000	1.35
3500	1.14	8500	1.37
4000	1.16	9000	1.40
4500	1.18	10000	1.45

HOW TO CALCULATE ACTUAL FAN PERFORMANCE AT OTHER THAN SEA LEVEL

A fan operating at an altitude above sea level is similar to a fan operating at air temperatures higher than 70°F.; it handles air less dense than standard. Table 2 gives the ratio of standard air density at sea level to densities of 70°F. air at other altitudes.

Example 3. Required: 5800 CFM at 6" SP at 5000 ft. altitude. 70°F. air at sea level weighs 1.20 times as much as 70°F. air at 5000 Ft. Therefore, at sea level, the SP is 1.2 x 6 = 7.20" SP. The fan would need to be selected for 5800 CFM at 7.2" SP at 70°F. .075 density.

When both heat and altitude are combined, the density of the air is modified by each, independently, so that the correction factors can be multiplied together.

Example 4. Required: 5800 CFM at 6" SP at 5000 ft. altitude at 600°F. Air at 70°F. at sea level weighs 2.00 x 1.20 = 2.40 times as much as 600°F. air at 5000 ft. altitude. At sea level and 70°F., SP = 2.40 x 6 = 14.4" SP. Select a fan for 5800 CFM at 14.4" SP. Divide the brake horsepower in the rating table by 2.40 to obtain horsepower at 600°F. and 5000 ft. If the fan is to start cold, it will still be at 5000 ft. altitude. Therefore, to get the “cold” horsepower requirement, divide by 1.20, the altitude factor only.

DENSITY CHANGES FROM OTHER THAN HEAT AND ALTITUDE

Fan densities may vary from standard for other reasons than heat and altitude. Moisture, gas, or mixtures of gases (other than air) are a few possibilities. In these cases, it is necessary to obtain the actual density of the airstream gas by some other reference material. A similar factor, as shown in Table 1, is then created using the standard density of air .075 lb. per cubic foot divided by the new density.

$$\text{Factor} = \frac{.075 \text{ lb./ft.}^3}{\text{special gas density}}$$

ACFM-SCFM DEFINITION

The terms ACFM and SCFM are often used in design work and cannot be used interchangeably.

SCFM is Standard Cubic Feet per Minute corrected to standard density conditions. To determine the SCFM of the volume used in Example 2, which was 11,000 CFM at 600°F., we would multiply the CFM by the density ratios.

$$11000 \times \frac{.037}{.075} = 5500 \text{ SCFM}$$

This indicates that if the weight of air at 600°F. were corrected to standard conditions its volume would be reduced to 5500 CFM

ACFM stands for Actual Cubic Feet per Minute. It is the volume of gas flowing through a system and is not dependent upon density.

The terms ACFM and SCFM are often used in system design work where both quantities need to be known. It should be remembered, however, that since a fan handles the same volume of air at any density, ACFM should be used when specifying and selecting a fan.

FAN SAFE SPEED AND TEMPERATURE

Whenever a fan is used to move air at temperatures substantially above or below 70°F., care must be taken to ensure that the safe speeds of wheel and shaft are not exceeded, and that bearing temperature and lubrication are satisfactory.

The maximum safe speed of a particular fan must be determined by calculations or actual tests. Safe speed depends entirely upon the wheel and shaft assembly's ability to withstand the centrifugal forces created by its own weight. Higher temperatures can affect the wheel and shaft assembly's ability to withstand these forces and therefore must be considered.

Most metals become weaker at higher temperatures. This weakness is measurable in terms of yield and creep strength. It can be translated into formulas that accurately determine the safe speed of a wheel and shaft assembly in relation to its tested maximum speed at standard conditions. Manufacturers provide safe speed reductions in their catalogs based on the alloy that was used to manufacture the wheel and/or shaft.

Some metals withstand heat better than others. Certain grades of stainless steel can be substituted to increase temperature limits. On the other hand, **fan wheels constructed of aluminum should never be operated above 200°F.**

For information regarding fiberglass reinforced plastic fan equipment, consult the appropriate product bulletin.

Table 3 gives an indication of the speed derate factors for several different alloys. These are listed for **reference purposes only.** For a specific fan, consult the appropriate product bulletin.

Table 3 - RPM Derate Factors By Material

Temperature °F.	Mild Steel	Aluminum	Stainless Steel		
			304L	316L	347
70	1.0	1.0	1.0	1.0	1.0
200	.97	.97	.88	.95	.95
300	.95	--	.82	.92	.93
400	.94	--	.78	.89	.90
500	.93	--	.75	.86	.90
600	.92	--	.73	.84	.90
800	.80	--	--	.79	.86
1000	--	--	--	.75	.83

The limiting temperature on any fan is the highest temperature that any component of the fan assembly will reach during any operating cycle. A fan in a process oven application may handle air several hundred degrees above the highest temperature the oven reaches, especially during start-up. On such applications, a temperature indicator should be located in the fan inlet to control the heat source and to keep the fan within its maximum safe temperature. This is particularly true where burners are located on the inlet side of the fan. In all cases, the fan should remain in operation until the air is cooled to 180°F. or less to prevent "heat soaking" of the fan shaft which could cause sagging.

Bearings must be kept cool; otherwise standard lubricants lose their effectiveness and bearing failures are likely. For axial fans, where the bearings are located in the airstream, care must be taken to ensure proper lubrication. Special fan and bearing designs, as well as high temperature lubricants, are available to extend the operating range to higher temperatures.

Arrangement 4 centrifugal fans, where the fan wheel is mounted on the motor shaft, should not be used above 180°F., unless special provisions are made (i.e., a shaft cooler or heat shield) to keep heat radiated from the housing from increasing motor bearing and winding temperatures.

When fan bearings are located outside of the airstream, as in Arrangement 1, 8, and 9 centrifugal fans, higher airstream temperatures are possible. Table 4 lists some typical maximum recommended operating temperatures for fans using ball or roller bearings.

A conventional fan using standard bearings and standard lubricant can normally be operated to a maximum of approximately 300°F. With the addition of a shaft cooler (Figure 3), this temperature limitation can be extended to 650°F. The shaft cooler has the effect of absorbing and dissipating heat from the shaft while circulating air over the inboard bearing.

Table 4 - Maximum Fan Inlet Temperatures

Arrangement 1 and 8 (Overhung Wheel)	
Standard Construction	300°F.
With Shaft Cooler	650°F.
With Shaft Cooler and Heat Gap	800°F.
With Shaft Cooler, Heat Gap, Stainless Wheel, and Alloy Shaft	1000°F.
Arrangement 3 (Wheel Suspended Between Bearings)	
Standard Construction	200°F.
Arrangement 4 (Wheel on Motor Shaft)	
Standard Construction	180°F.
Enclosed Bearing Fans (Axial Fans)	
Arrangement 4	105°F.
Arrangement 9	120°F.
With Special V-Belts with 2.0 S.F.	200°F.
Arrangement 9 Duct Fan With Heat-Fan Construction	600°F.
Plenum Fans	
Arrangement 3	105°F.
Arrangement 4	105°F.

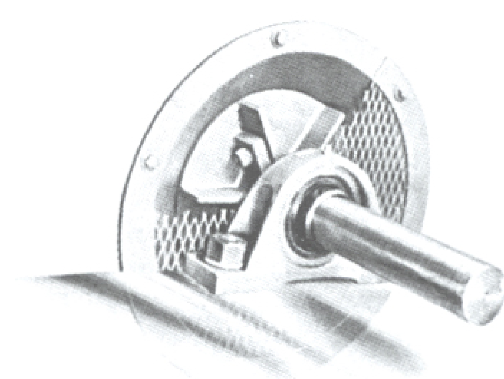

Figure 3 – Shaft Color

With the addition of a heat gap (Figure 4) the temperature limitation can be extended to 800°F. since the fan pedestal is isolated from the hot fan housing. For specific applications, consult the appropriate product bulletin. Also, recognize that these limitations apply only to bearings and that wheel and shaft limitations must be treated independently.

All of the foregoing is based on the use of standard lubricants. When high-temperature lubricants are required, the type of lubricant and the frequency of relubrication are normally much more critical.

When the fan shaft is heated to the point that it expands more than the structure to which it is attached, one expansion bearing and one fixed bearing should be furnished. The fixed bearing is located on the drive end of the fan while the floating bearing is located next to the fan. This arrangement, however, is not critical and may vary by manufacturer.

When the fan is handling air below 70°F., there is the possibility of other problems. Below -30 to -50°F., ordinary steel is too brittle. Aluminum wheels or wheels of steel containing at least 5% nickel must be used, and shafts must be made of nickel-bearing steel. In addition, lubricants become stiff, or even solid in these low-temperature applications. Exact operating conditions should be given to the fan manufacturer to relay to the bearing supplier for proper selection.

CALCULATING "HOT" RESISTANCE FOR SYSTEMS

Figure 5 shows a system that operates at the same temperature throughout. If the inlet temperature is known, the fan may be selected from the fan capacity tables and the rated horsepower and static pressure corrected by the temperature correction factor from Table 1. However, what happens to the system that the fan was operating against? If a fixed system, which originally was calculated for standard air, was subjected to the same temperature increase as the fan, then system static pressure will change and be identical to the fan static pressure change. The result is that if a fan and system operate together the flow will remain unchanged. (See Figure 6.) Unfortunately, this example assumes that the entire system is being subjected to the same temperature change, which is not always the case.

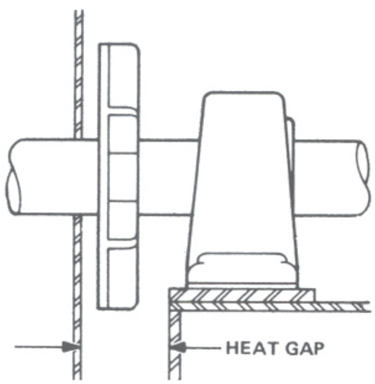

Figure 4 – "Heat Gap" between fan and bearing.

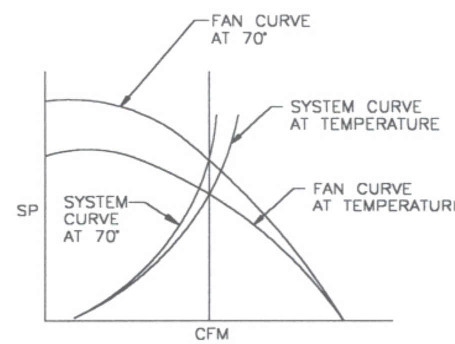

Figure 6 – Fan-system curve relationship with fan and system at the same temperature.

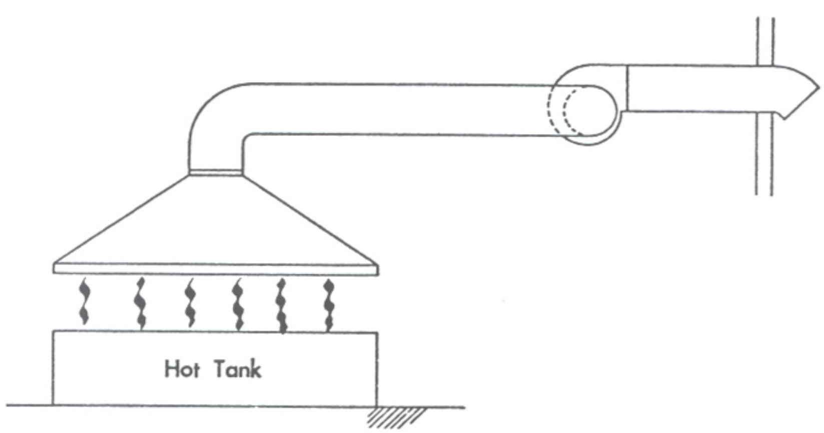

Figure 5 – A system with the same temperature throughout.

Figure 7 shows a system in which different temperatures are involved. The fan will not handle the same volume of air when operating hot as it does when cold. If the burner is on, the fan will handle 1430 ACFM against an actual static pressure of 1.2 inches. This is arrived at by adding the filter, burner, and nozzle resistance, neglecting for the sake of simplicity any external resistance from additional ductwork. The fan would be selected from the capacity tables on the basis of 1430 CFM at 1.72 inches static pressure (300°F. correction factor times 1.2 inches).

If the burner is turned off while the fan continues to operate at the same RPM, it is necessary to determine the system characteristic curve and plot its intersection with the fan to determine how much air the fan would move and at what static pressure. To accomplish this we must assume an arbitrary capacity, such as 1000 CFM at 70°F. The filter louver resistance would be the same, cold or hot, at .3 inches 70°F. The burner resistance would remain unchanged with temperature since it must be assumed that air expansion takes place after the high velocity section of the burner. The nozzles will vary in resistance directly as the density changes and inversely as the square of the flow. The nozzle would then have a resistance cold at 1000 CFM of:

$$.5'' \times \left(\frac{1000}{1430}\right)^2 \times 1.43 = .35''$$

Summing these resistances yields the cold resistance at 1000 CFM of 1 .05"SP. This new system point and corresponding curve are then plotted against a fan curve at standard conditions such that the resulting intersection will be the final operating point of the cold system. Using an actual fan as an example, the resulting flow would be 1220 CFM at 1.56 inches static pressure. (See Figure 8.)

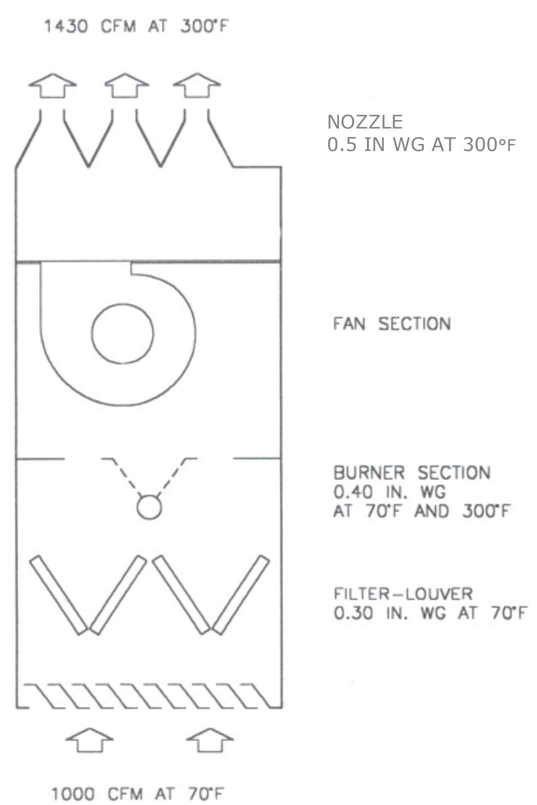

Figure 7 - A system with different temperatures.

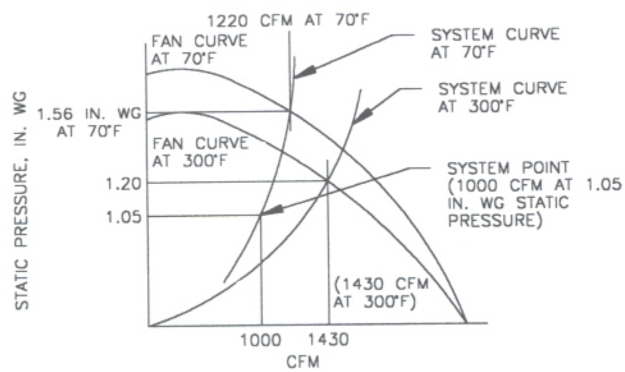

Figure 8 - Fan-system curve relationship with fan at different temperatures.

FAN LOCATION IN HOT PROCESS SYSTEMS

Figure 9 shows how a fan may be located more economically in one part of a system, as contrasted to another. Suppose 10,000 CFM is to be heated from 70°F. to 600°F. Obviously, the heater will require the same 3-inch pressure differential whether the fan is to push the air into, or pull the air out of, the heater.

A fan pushing air into the heater would be specified to handle 10,000 CFM at 70°F. against 3 inches of static pressure at 70°F. One possible selection is a fan with a 27-inch wheel diameter, Class I design utilizing a 7 1/2 HP motor.

The alternative fan, pulling air from the heater, would be specified to handle 20,000 ACFM at 600°F. against 3" SP at 600°F. It would be selected from the capacity tables for 20,000 CFM at 6" SP. One suitable choice is a fan with a 3 6 1/2-inch wheel diameter, Class II design utilizing a 15 HP motor. (Note: 26 HP, from the tables, at 70°F., divided by temperature correction factor, is 13 HP at 600°F.) This example illustrates why it is usually more economical to locate the fan at the coolest part of the system. In this case, the "push" fan might cost half as much as the "pull" fan.

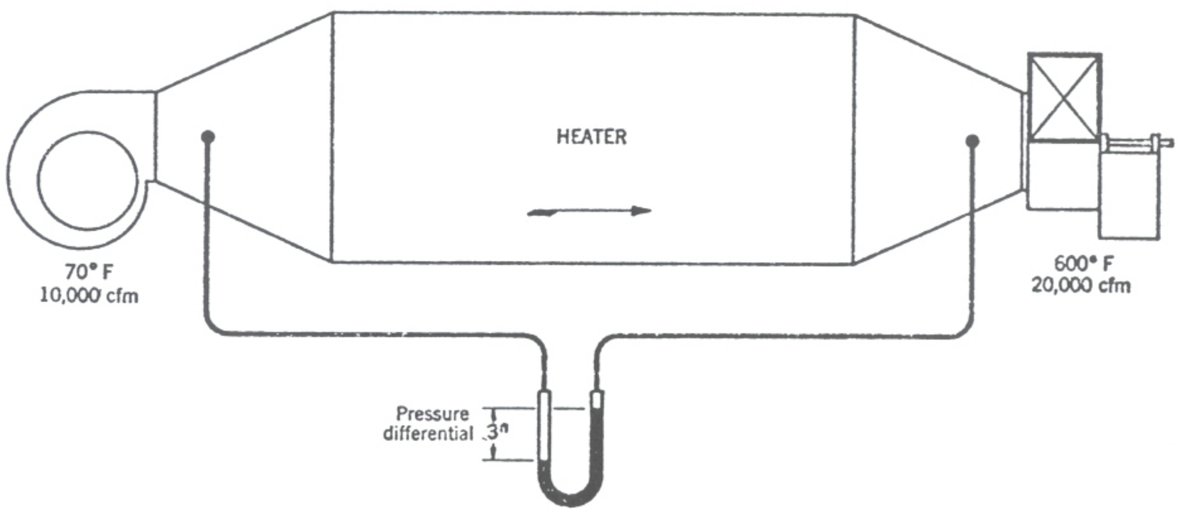

Figure 9 - The importance of fan location.